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Aeroacoustic Study of an Axial Ring Fan using Lattice-Boltzmann Simulations and the Ffowcs-Williams and Hawkings Analogy

Dominic Lallier-Daniels¹*, Mélanie Piellard², Bruno Coutty²*, Stéphane Moreau¹*

Abstract

The article presents the results of the acoustic study of a typical engine cooling fan module from the automotive industry with the help of lattice-Boltzmann simulations. The direct acoustic results obtained from simulation were compared to those from an experimental campaign, with the simulation mimicking the test conditions, and they showed good agreement with the measurements.

Furthermore, a study based on the Ffowcs Williams and Hawkings (FWH) analogy was carried out in order to identify the main contributors from the installation to the farfield noise footprint. To achieve this, a proper separation of the surface sources was carried out during the preprocessing stage of the simulations. It will be shown in this paper that accurate noise source identification can lead to simple in-situ modifications of a design.

Keywords

Low Speed Axial Fans — Aeroacoustics — Source Identification

INTRODUCTION

In the context of turbomachinery design, more and more emphasis is being put on the aeroacoustic performance of novel systems in addition to the constraints regarding aerodynamic efficiency. Designers must therefore take steps to be able to correctly predict the acoustic performance of the product and be able to identify the noise mechanisms involved to eventually control them. Achieving this through experiments in the early stages of design, while desirable, is often out of the question. On the other hand, in recent years numerical simulations (Computational Fluid Dynamics, CFD) have proven useful in realizing early detailed analysis of both the aerodynamic and aeroacoustic performance of turbomachinery systems, helping the designers to ascertain and optimize their design performance early in the development process.

The current paper presents the foundation of one such study using the numerical solver PowerFLOW to simulate a complete automotive radiator cooling module geometry (Condenser, Radiator and Fan Module, CRFM). This study is a continuation of the work done on similar geometries and shown in previous publications [1, 2].

The concurrent aerodynamic/aeroacoustic study is made possible by the use of lattice-Boltzmann simulations, which are inherently unsteady and compressible, allowing for direct acoustic simulation [3, 4, 5]. An experimental acoustic study was performed in a semi-anechoic environment in parallel with the simulations to provide data for the comparison and validation of numerical results.

The current paper aims at presenting the results of a comparison of the direct acoustic simulation with available experimental data. Further processing of the simulation results is carried out to identify and quantify the noise sources on the module. The analysis is done via the use of the Ffowcs-Williams and Hawkings analogy.

PRESENTATION OF THE NUMERICAL MODELS

Lattice-Boltzmann Method

The numerical code used for the simulation of the fluid flow through the fan module presented in this article is the Exa PowerFLOW software. In opposition to the more popular Navier-Stokes based solvers currently used in the industry, the software is based on the lattice-Boltzmann method and thus resolves the fluid flow at the so-called mesoscopic scale. The macroscopic quantities of the flowfield are then recovered using the appropriate moments.

The basic form of the continuous Boltzmann equation is presented as

\[
\frac{\partial f}{\partial t} + c_i \frac{\partial f}{\partial x_i} + \frac{F_i}{m} \frac{\partial f}{\partial c_i} = \left( \frac{\partial f}{\partial t} \right)_{\text{coll}} \tag{1}
\]

where \( f = f(\vec{x}, \vec{c}, t) \) is the mesoscopic scale particle distribution function at spatial coordinates \( \vec{x} \) and time \( t, \vec{c} \).
is the particle velocity, $F_i$ corresponds to external forces and $m$ is the molecular weight of the fluid. The term $\frac{\partial f_i}{\partial t} + \mathbf{c_i} \cdot \nabla f_i$ in Eq. 1 is the collision term describing the interaction of the particles. The collision operator, describing elastic collisions between the particles, must therefore satisfy the conservation of mass, momentum and kinetic energy during the collision itself. Using a collision operator called the BGK operator (Bhatnagar, Gross and Krook, [6]) and a Chapman-Enskog [7] development, Eq. 1 recovers the Navier-Stokes equations.

The Boltzmann equation can also be discretized to allow for only a subset of velocity vectors $\alpha$ to be applied to the particles [8] on a cubic lattice. The PowerFLOW software uses a 19 discrete velocity model [9]. This, however, restricts the validity of the lattice-Boltzmann method to low Mach number flows (typically $M=0.4-0.5$ is cited as the limit). The discretized form of Eq. 1 in both space and time using the BGK model is written as

$$f_\alpha(x_i + c_\alpha \Delta t, t + \Delta t) - f_\alpha(x_i, t) = -\frac{1}{\tau} [f_\alpha(x_i, t) - f^{eq}_\alpha(x_i, t)] \tag{2}$$

For simulations involving rotating parts, the computational domain can be made to include regions using a non-inertial rotating reference frame [10, 11]. Data is exchanged between the inertial and non-inertial regions using a suitable interface. In the non-inertial region calculations, a pseudo body force $F_\alpha$ is introduced in Eq. 2. This body force $F_\alpha$ corresponds to the Coriolis, centrifugal and Euler pseudo-forces normally found in a rotating non-inertial reference frame.

The code PowerFLOW also utilizes a modified relaxation time $\tau \rightarrow \tau_{turb}$ derived from the renormalization group $k - \epsilon$ transport equations ([12, 13]) in order to model the unresolved turbulent small scales in the LBM calculations. This approach is referred to as the LBM very large eddy simulation (VLES).

The LBM possesses some key advantages over traditional CFD methods. The LBM is inherently an unsteady and compressible method. This is of importance when trying to study fundamentally transient phenomena such as in turbomachinery applications. Moreover, while the LBM is globally 2nd order accurate, it shows acoustic dissipation properties equivalent to 6th order centered Navier-Stokes finite difference schemes and dispersion equivalent 2nd or 3rd order schemes [4, 5, 3]. This allows for direct acoustic evaluation in the farfield. The basic discretized Boltzmann equation (Eq. 2) being explicit, the LBM calculations can be efficiently implemented in a parallel computing environment. Using modern parallel computing clusters, large and complex geometries can thus be simulated with a significant amount of physical time with fast turnaround.

**Ffowcs Williams and Hawkings Analogy**

In the PowerFLOW suite is included a farfield analysis module that allows for the use of the FWH analogy [14] based on Farassat’s 1A formulation of the FWH equation [15, 16, 17, 18]. The formulation supposes a permeable surface $f(x, t) = 0$ with outward surface normal $\hat{n} = \nabla f$. The surface encloses a volume $V$ defined by $f(x, t) < 0$. The surface is defined as having a certain velocity in 3D space.

For the solid formulation, the Exa software suite uses, as mentioned before, Farassat’s 1A formulation of the FWH equation. The complete development is quite lengthy and not reproduced here; the reader is directed to Farassat’s and Brentner’s work for full reference [15, 16, 17, 18]. The formulation is concerned with the calculation of the acoustic pressure radiated from the source region. It uses a free-field Green’s function for a quiescent medium to solve the wave equations for the monopolar and dipolar terms whilst neglecting the quadrupolar source term from the FWH equation. The acoustic pressure $p'$ is thus given at an observer position $x$ from the source terms emanating at position $y$ from the monopolar (thickness, $T$) and dipolar (loading, $L$) terms:

$$p'(x, t) = p'_T(x, t) + p'_L(x, t) \tag{3}$$

The Exa software suite also includes a second formulation of the FWH equation which is suited for example for wind tunnel applications wherein both observer and source are stationary but immersed in a uniformly moving medium. The formulation can be recovered by redeveloping the original FWH equation [19] or by retaking formulation 1A but considering now the observer and source to be moving at constant speed in a quiescent medium, modifying the thickness and loading terms as well as the radiative distance [20].

The source terms are evaluated using an advanced time approach [14, 21], taking advantage of the regularly sampled numerical data measured at $t_{ret}$. The acoustic pressure is then propagated into the farfield to designated observer points at a time $t$. This will likely result in unevenly spaced observer time; this is resolved by using interpolation of the given pressure signal at the observer to restore the received signal sampling rate to that of the source term.

The two FWH analogy formulations available in the Exa software allow for the use of both solid surfaces moving in a quiescent medium or stationary surfaces (permeable or solid) in a uniformly moving medium as sources for the propagation of farfield noise.

**SIMULATION SETUP AND AVAILABLE DATA**

The geometry studied in this paper consists of a complete engine cooling module constituted of a fan, an electrical motor, a shroud as well as a series of heat exchangers. The simulated geometry is shown in Fig. 1. Due to the complexity of the grid refinement that would be required to properly mesh the heat exchangers, they were modeled in the simulation using porous media regions. The inertial and viscous resistance of the media were set in such a way that it reproduced the pressure loss incurred in the actual heat exchangers. A blockage plate located downstream of the module (outlined in Fig. 1) is also modeled in the simulation in order to reproduce the proximity of the engine block when mounted on a car. The electric motor driving the fan was also modeled.
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In order to reproduce the semi-anechoic environment from the experiment conducted in parallel, the modeled CRFM was inserted in a large simulation domain, as schematically shown in Fig. 2. The floor is in solid grey while all other outer surfaces have free-flow conditions. High viscosity is artificially imposed in the outer layers of the fluid domain to prevent the reflection of acoustic waves back to the source.

**Figure 1.** Illustration of the studied fan module

**Figure 2.** Illustration of the semi-anechoic simulation domain. [2]

In the simulation, as in the experiment, the CRFM is operating at free-flow conditions (uniform pressure in the room) with the fan rotational speed imposed at 2535 RPM.

A series of 20 probes located 0.5 m upstream of the CRFM was included in the simulation setup to mimic the experimental microphone array used to collect acoustic data in the semi-anechoic chamber (Fig.3). Note the presence of the downstream blockage plate in the experimental setup on the right side of Fig. 3.

Finally, the installation effects due to the microphone array in the experiment are not represented in the simulation.

**SIMULATION RESULTS**

In the course of this study, the direct acoustic signals recorded in the simulation have been compared with the experimental data gathered in the semi-anechoic room through the aforementioned microphone array. As an example, two comparisons for different microphones are shown in Fig. 4. The abscissa shows the frequency normalized by the rotational frequency of the fan, referred to as order (abbreviated by “O”). The frequency resolution is set to 10 Hz.

**Figure 3.** Illustration of the experimental setup in the semi-anechoic room for the CRFM. Flow from left to right. [2]

The first numerical spectra are extracted from a simulation with only 0.35 s of physical time available (termed Sim 1). The second simulation had 1 s of available physical time (termed Sim 2). Both simulations have identical geometries, but the second simulation also has a more detailed surface decomposition for the FWH analogy propagation for source identification. This is detailed in the next section.

Overall, a good correlation between the experiment and simulation is obtained, with some variance between the different evaluation locations. A better agreement with the experimental spectra is also obtained with the longer simulation; this is not particularly surprising as it allows for the use of longer steady-state time data for use in the Fourier transform, which is carried out using Welch’s periodogram method.

Furthermore, the spectra show that the configuration produces consistent tonal noise around the first Blade Passing Frequency (BPF) at O7, with the level varying with the considered microphone, indicating an uneven directivity pattern, which is to be expected. There is also a large broadband hump in the frequency range at and below the second BPF (O12-14). This can be attributed to the presence of rotating turbulent structures in the flow interacting with the fan blades, causing sub-harmonic humps. This phenomenon was discussed in previous publications for similar geometries [2, 22, 23, 24]. The third BPF also emerges as a discrete tone, although again its emergence level differs from microphone to microphone. A discrete tone at 760 Hz (O18) is also observed in the measured spectra but is not always well represented by the simulation. Its amplitude varies from microphone to microphone in the experiment. The origin of this particular tone was not pinpointed in the experiment.

Given that the heat exchanger, modeled by porous media in the simulation, acts like a flow conditioner, the inflow should be relatively free of any large distortions from upstream. However, performing an analysis of the flow field in the volume surrounding the fan using visualizations of the $A_2$ factor [25]
Figure 4. Comparison of the acoustic pressure PSD from the direct CAA with the experiment for two separate microphone locations (a) Microphone 2 (b) Microphone 14 in Fig. 5 highlights the presence of several characteristic patterns in the vortical field. First of all, the appearance of an annulus of structures of varying shapes and sizes is seen close to the fan ring, originating from the tip gap and wrapping around it towards the tip of the blades. Larger, more radially penetrating structures are seen appearing at locations where the CRFM frame is closest to the fan along the circumference. This occurrence might be linked to the appearance of strong BPF harmonics as it creates static disturbances in the flowfield at regular intervals. These structures' influence on the three components of the velocity field can be seen in Fig. 6 in a plane just upstream of the fan.

A string of coherent vortical structures originating near the trailing edge of the blade at 70% span is also seen appearing. A ring of vortical structures near the hub at the root of the blades is also observed. Generally, the structures identified are seen to be rotating at a fraction of the fan speed and thus would interact with the following blades and lead to an increased generation of tonal and broadband noise.

The filtered pressure fluctuations on the surface of the fan, stator and motor elements of the CRFM were also analyzed to try and understand of the possible sources of high tonal noise in the frequency bands identified on the spectra. Fig. 7 shows the PSD of pressure fluctuations for a frequency range corresponding to the 1st BPF.

It can be observed that the main areas of high pressure fluctuations are located on the fan blades and stator arms, with generally more elevated levels near the tip. In the case of the fan blades, there are specific areas where high pressure levels can be seen, such as on the suction side of the blades near the leading edge at the hub and tip of the blades, as well as near the trailing edge near the fan ring and at approximately 70% of the span. These locations correspond well to the appearance of vortical structures in Fig. 5.

However, while the filtered wall pressure levels provide a certain indication of the probable origin of the noise, they do not necessarily represent areas which contribute to the farfield noise (e.g. evanescent sources, destructive or constructive interaction between sources during propagation). One way to ascertain the source of the noise is to employ a hybrid method in order to propagate these pressure fluctuations into the farfield, done here via the use of the Ffowcs Williams and Hawkings analogy. This allows to identify and quantify the different sources and their contribution to the noise spectra. The studied geometry was decomposed in a way as to allow for a precise identification and localisation of the dominant noise sources using this method. The detailed analysis is the subject of the next section.

SOURCE IDENTIFICATION USING THE FFOWCS WILLIAMS AND HAWKINGS ANALOGY

In order to properly identify and evaluate the primary sources of noise contributing to the farfield spectra, the Ffowcs Williams
Figure 6. Instantaneous velocity field directly upstream of the fan (a) Azimuthal velocity normalized by the relative tip speed (b) Axial velocity (c) Radial velocity

Figure 7. PSD of wall-pressure fluctuations on the fan, stator and electrical motor surfaces for the 232-358 Hz frequency range (O12-14)

and Hawkings (FWH) analogy, which was presented previously, was used to propagate the surface pressure fluctuations
to farfield locations. The solid surfaces for which pressure data was recorded in both simulation cases studied here are identified in Fig. 8. They consist of the fan blades (orange) and hub (magenta), fan ring (blue), shroud ring (purple), stator (green) as well as the engine stator (red) and rotor (cyan, not shown here). The pressure fluctuations on the module frame (white) and restriction plate located behind the fan module were also recorded in the longer simulation case.

Figure 8. Illustration of the separation of the surface sources on the fan module (a) View from upstream (b) View from downstream

In order to provide a basis for comparison of the FWH analogy with both the direct acoustic simulation data as well as the experimental measurements, the primary observer locations were chosen to correspond to the aforementioned microphone locations (see Fig. 3).

A second series of observer locations was also setup to evaluate the acoustic directivity of the simulated fan module, as no directivity measurements were carried out experimentally. The observer locations were defined on two 180° arcs located upstream of the fan system in both the horizontal and vertical planes, as illustrated in Fig. 9. The arc radius was set to 1.5 m.

Figure 9. Illustration of the evaluation arcs for directivity (a) Horizontal plane (b) Vertical Plane

FWH Results on Short Simulation
The FWH analogy was first applied to the first simulation (Sim 1), which provided some insight that led to improvements in the setup of the second simulation (Sim 2).

The results from the FWH propagation to the microphone array observer locations, which allows for the comparison of the experimental measurement data with the simulation, are presented in Fig. 10 for two microphone locations. Two FWH prediction curves are presented, which differ with regard to the propagation model used: the first curve represents the noise levels predicted by the FWH analogy using a fully free-field propagation method, whereas the second one takes into account the acoustic reflection from the solid floor in close proximity to the fan module.

Figure 10. Acoustic pressure PSD for two separate microphone locations in the first simulation comparing the FWH solid formulation with and without the effect of ground reflection (a) Microphone 2 (b) Microphone 14

From Fig. 10, it can be observed that the PSD levels predicted by the analogy are generally 5-10 dBs lower than those yielded by the experiment and direct CAA. It was surmised that the missing sources (module frame and restriction plate) could be responsible for at least a part of the observed
difference, which is why they were included in the second simulation. The added effect of the acoustic reflections due to the floor also seems to lead to some improvement in the prediction of the levels of the first and third BPF tones when compared to the direct probe measurements for the presented microphone locations.

The effect of the acoustic reflection on the FWH analogy is more readily observed when looking at the directivity patterns upstream of the fan module. Fig. 11 shows the acoustic directivity in the horizontal and vertical planes obtained from the simulation data. The direct acoustic probe results are compared with the analogy. The data from the horizontal arc is presented in the top half of the polar graphs (0-180° from left to right in Fig. 9(a)) while the vertical directivity pattern is shown in the lower half (0-180° top to bottom in Fig. 9(b)).

Overall, the FWH curves with reflection modeled correlate better with the direct CAA, both in level and shape. The effect is greatest for the 2nd and 3rd BPFs as well as O18 (higher frequency range) in the vertical plane, where the lobed patterns exhibited by the CAA are much better represented by the FWH analogy when including the acoustic reflection. Thus, the inclusion of relevant reflecting surfaces in the propagation model leads to a much better representation overall as indicated by the directivity patterns.

However, there are still gaps between the direct numerical acoustic evaluation and the FWH analogy. As mentioned before, several sources were not recorded in the original simulation, especially the module frame and blockage plate, which are expected to produce elevated levels of wall-pressure fluctuations as a result of their interaction with the flow around the fan. They were thus included in a second, more detailed, simulation in order to evaluate this hypothesis.

**FWH Results on Long Simulation**

Following the observations made with regard to the effect of the acoustic reflection modeling on the short simulation, it was included directly in the FWH analogy when applied to the second simulation (i.e. freefield propagation is not included for the second simulation). The module frame and blockage plate surfaces were also added to the list of recorded sources. In addition, a permeable surface englobing both the fan module and the downstream blockage plate was added in the simulation to test the permeable FWH formulation. The aim was to investigate whether significant noise sources were present in the flow itself or if the majority of the noise was produced as a result of fluid-solid interactions.

The result of the FWH prediction, including ground reflection effects, for the second simulation is shown in Fig. 12 for two of the microphone array’s observer locations. The primary conclusion that can be drawn is that the FWH predicted noise levels follow the direct CAA very accurately, indicating that the majority of the noise sources have been captured. There is also little to no difference between the solid and permeable surface formulations of the analogy for this case, indicating that the main sources of noise are indeed the result from fluid-solid interactions.

The directivity patterns as predicted by the simulated direct acoustics and the FWH analogy are also presented in Fig. 13 for the second setup in a manner similar to the previous section. A good overall agreement between the FWH results and direct CAA measurements is found in both level and shape for the directivity patterns. This is especially true for the 2nd BPF and the O18 frequencies, where the rather complex lobes predicted by the direct CAA are well reproduced by the FWH prediction in both level and shape. For the 1st and 3rd BPF however, while the direct CAA directivity pattern are well represented by the FWH analogy, a significant difference in the PSD levels is observed (8 dB for the 1st BPF and 10 dB for the 3rd BPF).

When comparing the current results with those obtained from the first simulation, the directivity patterns predicted by the FWH analogy are indeed much closer to the direct CAA. This indicates that the dominant noise sources are well captured by the FWH analogy. The main interest of using the FWH analogy in this case lies in the identification of the dominant noise sources. To achieve this the surfaces for which the wall-pressure fluctuations were recorded were separated to allow for the identification of the source of the radiated farfield noise. Fig. 14 shows the detail of the noise spectra for an array microphone location by surface of origin. The sources were grouped into three categories for readability purposes: rotor elements, stator elements and motor. The restriction backplate is also included in the figure.

One of the first things that can be observed is that the blockage plate exhibits a prominent PSD level across most of the frequency spectrum. This is due to the fact that the plate is exposed to the perturbed exhaust flow from the fan, causing high pressure fluctuations on the surface which are then propagated into the farfield as noise. The backplate contribution is thus correlated with other sources on the fan. A cross-correlation analysis would be required to investigate further, but exceeds the scope of the current paper.

The PSD levels from the stator elements also follow almost exactly the levels from the module frame, which is included in this category. This provides an explanation to the fact that the FWH prediction matches the CAA data much better in the second simulation when comparing to the first case, as a dominant noise source was omitted in the evaluation for simulation 1, leading to decreased overall PSD levels in the spectra.

Taking a closer look at the three first BPF harmonics as well as the O18 tone highlighted earlier, the source elements can be graded against each other and some insight can be gained on the possible source mechanisms.

First of all, for the first BPF tone, the rotor and stator elements appear as primary contributors. The flow non-uniformities in the velocity field near the top and bottom parts of the module frame (see Fig. 6) would in fact favor periodic fluctuations on the rotor blades near the first BPF and thus the production of the associated tone. It is surprising however that the stator elements (stator arms precisely) sometimes surpass the rotor elements as the primary contributor for this tone.
This happens on microphone 2, which is the one shown in Fig. 14, but also on other microphones in the array. No explanation was found for this behavior.

The second BPF frequency range spanning O10 to O14 also shows an interesting trend. It can be observed that the stator elements contribute to the second BPF where a peak is shown, but there is also a large subharmonic hump centered around O12-13. However, the rotor elements show no tonal peak at the second BPF, but rather a subharmonic hump around O12 is clearly visible. This indicates that the second order BPF tone originates exclusively from the stator and that the rotor does not contribute significantly. The subharmonic hump mechanism for the rotor, however, is surmised to be due to the influence of the vortical structures primarily issued from the tip gap region and highlighted in other publications [2, 22, 23, 24] on similar ring-fan geometries.

For the third harmonic of the BPF, the rotor and stator elements are shown to be the dominant noise sources, although like the first BPF the dominant source can change depending on the microphone considered. Of interest here also is that the motor elements also show a significant contribution to this tonal peak, with the level on microphone 2 from Fig. 14 still being approximately 10 dB lower than the rotor and stator contribution. However, once again, depending on the microphone being evaluated, motor elements can produce a tonal peak whose level equals or exceeds that of the rotor and stator.

A significantly high tonal peak is also seen emerging
from the motor contribution at O18. Tonal noise at this frequency was also observed consistently in the experimental measurements but the source could not be identified. The FWH analysis would however point to the motor assembly as the culprit. Observing the interior of the fan hub in Fig. 15, a series of 18 regular ‘ribs’ can be identified. The motor is also mounted using three protruding anchor points sitting a few millimeters from the ribs. The interaction of those two elements during fan rotation could very well lead to an O18 tone. The O18 tone was in fact shown to be due to these geometrical features through experimental measurements on the module: when the hub ‘ribs’ and motor mount were concealed, the O18 tone all but disappeared. The study is not detailed in the paper for conciseness.

It should be noted that the independent Fourier decomposition of the different sources results in the loss of any information as to the the correlation between them. To delve deeper into the issue, a cross-correlation analysis for a fixed observer position is required, but is not shown here as it exceeds the scope of the present study.

CONCLUSION

This paper presented the details of LBM simulations carried out on an engine cooling fan module typical of those used in the automotive industry. As highlighted in previous publications on similar engine cooling module geometries [1, 2] and reproduced here, the comparison of the simulated acoustics with those measured on the physical setup representative of the simulated geometry in a concurrent experiment shows a good agreement, thus confirming the capability of the LBM simulations at reproducing the acoustics of a fan system included in a rather complex installation.

Furthermore, the use of the FWH analogy using recorded wall-pressure fluctuations from the module solid surfaces, combined with an extensive decomposition of the surfaces, allows for the identification and localization of noise sources on the module. This method also allows for the evaluation of their contribution to the farfield noise spectra. The results obtained from the analogy also highlight the limits of using visual methods such as PSD maps of wall-pressure fluctuations to identify noise sources.

The caveats of using the FWH analogy were also presented, with the investigation yielding as a byproduct indications that care must be taken to include all potential noise-generating surfaces. Furthermore, the effects of reflection for the farfield propagation must be taken into account for the method to yield relevant results.

The usefulness of the FWH-based methodology presented here showed its value by allowing the use of the numerical results to identify the source of an O18 tone whose source had not been identified using experimental data alone. This led to simple modifications on the fan module which lowered the overall acoustic footprint of the system.

REFERENCES


Figure 13. Comparison of the acoustic directivity as predicted by direct CAA and the FWH analogy for the second simulation (a) 1st BPF (b) 2nd BPF (c) O18 tone (d) 3rd BPF


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Figure 14. Decomposition of the contributions of the different surfaces sources for microphone 2

Figure 15. Illustration of the interior of the fan hub.


